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TWIN SCREW MACHINES TO REPLACE THROTTLE VALVES IN REFRIGERATION SYSTEMS

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ABSTRACT

As the component efficiencies of vapour compression refrigeration systems approach their upper limits, the losses due to throttling within the cycle become more significant, especially with the newer refrigerants. Two-phase expanders to replace the throttle valve and recover power from the loss of the throttling process then become more attractive. The use of twin screw machines for this purpose is considered with the power so recovered to be used in a variety of ways. These include direct drive of the main compressor, an electric generator, or another compressor or direct recompression of part of the vapour formed during expansion within the same pair of rotors.

INTRODUCTION

Power recovery from the throttling process in vapour compression cycles is well known as a potential means of improving refrigeration and air conditioning system performance. However, with potential power recovery of less than 100kW in all but the largest systems, to date no cost effective method has been identified for achieving this.

In recent years, the need to maintain or improve the Coefficient of Performance with new refrigerants such as R134a has led to increased efforts by system manufacturers to raise compressor efficiencies and reduce temperature differentials in the heat exchangers. R134a has advantages as a refrigerant, in that it has no atmospheric ozone depleting tendencies. However, due to relatively large throttling losses, its use in chiller applications results in a reduction in the C.O.P of some 6% relative to R11. As already shown, (Brasz 1999) this disadvantage can be largely overcome by replacing the throttle valve by a turbo expander, the output of which can be used to reduce the compressor work input. However, the installed cost of such a component is very high.

Various investigators, such as.Taniguchi et al, 1983 and Smith and Aldis, 1990, have examined the use of a twin-screw expander as a throttle valve replacement.

The main reason for its non-adoption up till now was high cost. The most significant factors responsible for this were poor adiabatic efficiency together with high manufacturing and installation costs. This work has been reviewed by Smith, 1999.

In positive displacement machines, the built-in volume ratio is defined as the ratio of the displacement volume at the opening of the discharge port to its value at the point of closure of the suction port. Recent studies by the authors (Smith and Stosic, 1994 and 1998 and Smith et al, 1999) have shown that, high adiabatic efficiencies are possible if the built in volume ratio should is much less than the overall volume ratio of the actual expansion process.

Another requirement for high adiabatic efficiencies is the need to maintain thermodynamic equilibrium between the liquid and vapour during the two-phase expansion process. The presence of lubricating oil in the working fluid inhibits this. Accordingly all early screw expanders needed timing gear, to avoid rotor contact, and internal shaft seals to separate the working fluid from the lubricant. The unsupported length of the rotors had to be increased to allow for the presence of these seals and this in turn led to the need for thicker rotor shafts and a larger casing. All these features raised the manufacturing cost. Moreover, seals which were acceptable in vapour or gas compressors were found to be inadequate in expanders because the refrigerant in the expander working chamber, which has a liquid content of approximately 80% by mass, absorbed the lubricating oil far more readily.

As shown in Fig. 1, various modes of installing the expander in the system have been proposed. These include; coupling the expander to an electrical generator, coupling the expander directly to the main compressor drive shaft or coupling the expander and a separate screw compressor together in a self contained sealed unit. More recently attempts have been made to develop a self contained unit in which a single pair of matching rotors both expands the two-phase fluid and recompresses part of the vapour so formed.

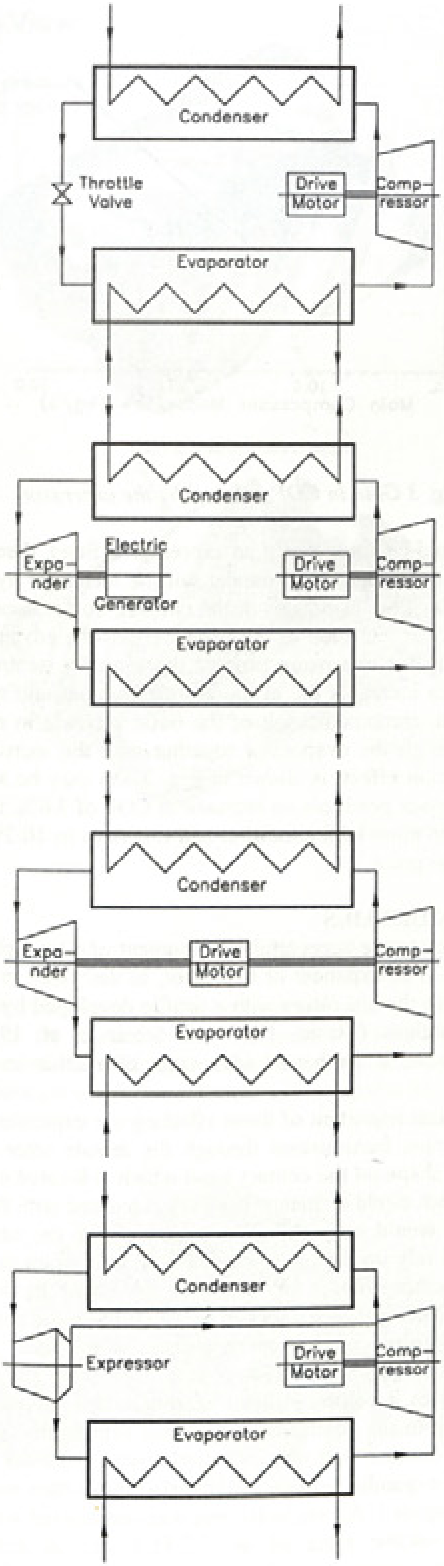

Fig. 1 Chiller unit with two-phase expander as alternative to the throttle valve

Electrical power generation involves the added cost of the generator. Also, the power generated undergoes two additional energy transformations before it can be utilised to reduce the power input to the main compressor. Linking the expander directly to the main compressor drive is superficially far more attractive but in fact is surprisingly expensive. This is partly because large bore piping is needed to route the condenser discharge to a position close to the main compressor and then reroute the expanded fluid back to the evaporator inlet with low pressure losses if the expander and compressor are to be close together. Both these arrangements require a control system to be built into the expander in order to keep its speed constant at part load. Also, if the expander is a turbine, a further control device is needed to avoid hot gas flashing from the condenser to the evaporator during start up.

In 1995 two of the authors (Smith and Stosic) proposed linking a twin screw expander to a twin screw compressor in a sealed unit to form an independent free running device. The expander would replace the throttle valve and the power, thus recovered, would recompress a portion of the vapour formed during the expansion process to deliver it directly to the condenser inlet. A key feature of the study was the demonstration that such a combined unit was capable of stable operation over its entire operating range from start-up to any required operating speed. Effectively it constituted two machines in one casing. To emphasise the combination of expansion and compression processes in the same unit it was called an expressor.

The expressor could be located directly between the condenser exit and the evaporator inlet, the seal between the rotor lobes would prevent hot gas flashing during start up and the speed of such a unit could vary according to the chiller unit duty without the need for additional controls.

ANALYSIS

A study was carried out on how effectively twin screw machines would recover power from the throttling process in a typical large industrial chiller unit with a centrifugal compressor and using Refrigerant R134A as the working fluid. Data on the chiller performance characteristics were supplied by a commercial manufacturer. A range of operating conditions was considered for each inlet guide vane angle with successive settings which covered the following conditions: evaporation temperature: 4.0-5.7 °C, condensing temperature: 29.4-40.3 °C refrigerant mass flow rates: 6.8-31.7 kg/s.

The method of analysis employed followed the approach which was developed successfully for the estimation of screw compressor performance by Hanjalic and Stosic, 1997, and is given in more detail by Smith et al, 1995. The geometric relationships which describe the rotor profiles were used to define an instantaneous control volume formed by the space between a matching pair of rotor lobes and the casing at any rotational position. This definition included the trapped volume, the inlet and exit flow areas and the leakage path areas between the engaging rotors and between the rotors and the casing. The equations of conservation of mass, momentum and energy were then applied to the instantaneous flow of fluid through this control volume. An equation of state was used to relate the thermodynamic properties of the working fluid passing through the expander. The use of these equations and the assumption of a rotational velocity led to a set of simultaneous non-linear differential equations whose instantaneous solution resulted in the estimation of the pressure within the trapped volume at any angle of rotation. The equations were solved by numerical integration describing the admission, expansion and discharge processes within the expander and estimates of the pressure-volume changes within it from which the mass flow rate, power output and adiabatic efficiency of the machine were readily derived.

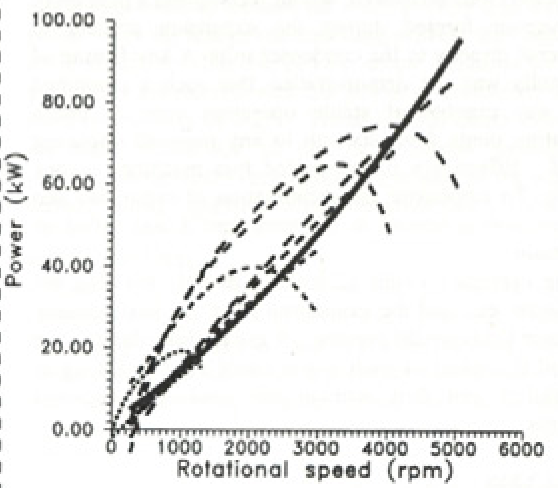

Fig. 2 Matched compressor-expander characteristics

To show how an expressor unit would perform over the chiller operating range, the superimposed expander and compressor characteristics for this arrangement are drawn in Fig 2. The operating line shown is obtained from the intersection of the two sets of curves. It may be seen that the expander will be able to drive the compressor as the mass flow increases at speeds up to 4,200 rpm under all chiller operating conditions. Moreover, since the operating line of the expressor lies on the negative slope of the compressor characteristic curves, the unit will always be stable in operation under conditions of changing load.

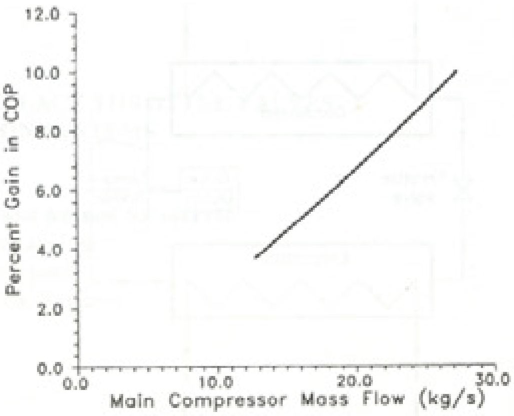

Fig. 3 Gain in COP induced by the expressor

It should be noted that if an expressor is fitted, then the mass flow through the expander will be increased by the extra mass flow it induces in the coupled compressor. In addition, the inclusion of the expander has the advantage of making the expansion process more nearly isentropic and hence increases the evaporation effect per unit mass flow. The combined result of the basic increase in mass flow through the evaporator together with the increased refrigeration effect, is shown in Fig. 3. As may be seen, the expressor produces an increase in COP of 3.6%, even at the minimum load condition and this rises to 10.3% at the design point

DESIGN DETAILS

The key to the successful development of a twin screw machine as an expander or expressor, as described in this paper, was the use rotors with a profile developed by one of the authors (Stosic, 1997 and Stosic et al, 1997). These confer a number of advantages over other known types.

The most important of these affecting the expander are small torque transmission through the female rotor and involute shape on the contact band which is located close to the pitch circle diameter. It was decided that with these rotors it would be possible to dispense with the timing gear and rely on the presence of liquid refrigerant to act as a coolant should any heat be generated by rotor contact. This was first checked by using the same profile rotors in an oil injected compressor in which water was used as the injected fluid (Stosic et al, 1998).

Advances in rolling element bearing design (Jakobson, 1996) led to the decision to use liquid refrigerant as the bearing coolant and thereby completely eliminate oil from the expander system. Internal shaft seals then would not be required. An expander unit was constructed with a built-in volume ratio of only 2.85:1 for an overall volumetric expansion of 11.4:1 including all these features and tested by the authors (Smith et al 1999).

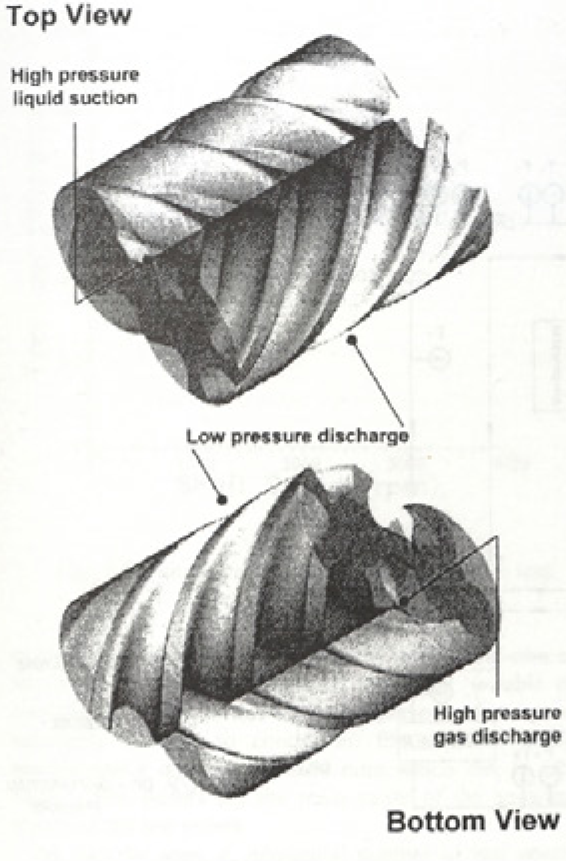

Fig. 4 Operating principle of a two-rotor expressor

The next stage of the programme was to determine whether the expansion and compression processes could be carried out efficiently in the same set of rotors. The principle on which this is based is illustrated in Fig 4.

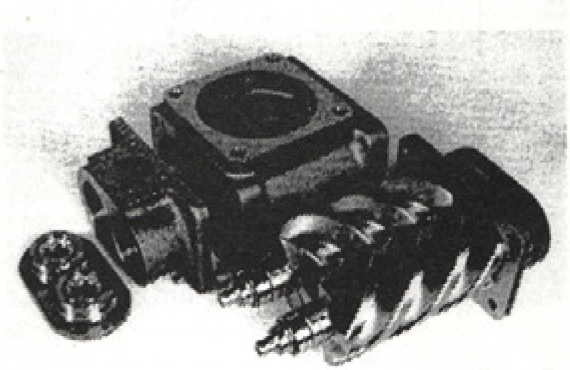

Fig. 5 The twin rotor expressor

Two problems associated with it were the need for a sealing line between the rotors on both contact faces and a large enough port in the casing to enable virtually all the liquid to be discharged at the end of expansion before recompression of the residual vapour. By increasing the rotor wrap angle to over 400° it was possible to maintain the trapped volume between the rotors and the casing virtually constant over a relatively large angle of rotation. A discharge port could then be formed in the casing over the whole of this region without adversely affecting either the expansion or compression processes. The final form of the twin rotor expressor unit as built and tested is shown in Fig 5.

EXPERIMENTAL DETERMINATION OF THE EXPANDER AND EXPRESSOR PERFORMANCE

Both the expander and expressor were tested on the same rig which was specially designed for two-phase expander development. The expander and expressor are shown as installed in Fig. 6 and the arrangement of the test rig is shown in Fig. 7. Note that the expressor and expander form parallel loops in the same circuit. Either loop could be operated independently but not simultaneously. A more detailed description of the system is given by Smith et al, 1996. Essentially, it removes the need for the large power inputs associated with the compression process of a vapour compression cycle system by using a high boiling point working fluid (R113). The refrigerant is pressurised by a liquid feed pump, heated to its boiling point only, without evaporation, expanded as a two-phase fluid and then condensed at above atmospheric temperature. A parallel loop enables the fluid to be circulated with the expander bypassed and this is used both for setting up the required temperature, pressure and flow conditions and as a safety feature in the event of expander failure.

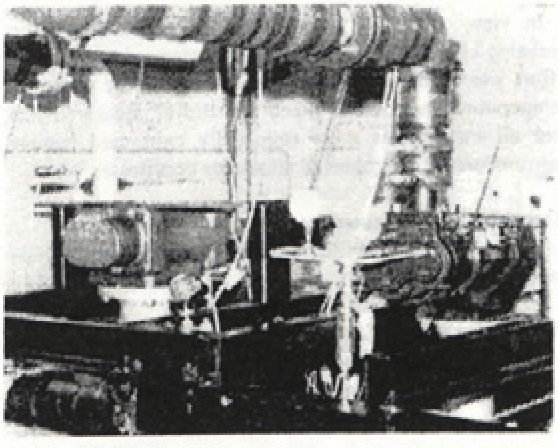

Fig. 6 A view of the expressor (left) and expander (right) installed in the test rig

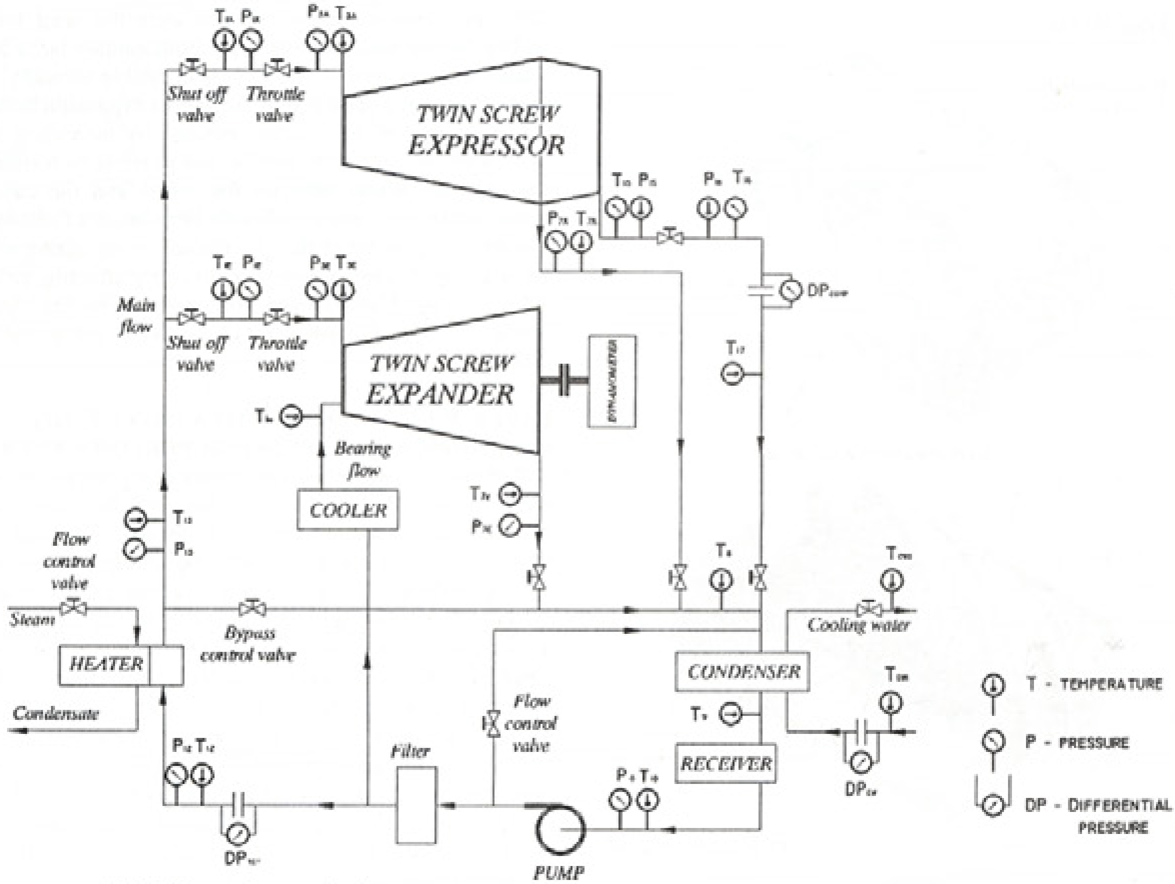

Fig. 7 Expander test rig layout

The rig was originally designed and constructed in 1984 and before use on the machines described in this paper, it was extensively refurbished, especially with regard to the data acquisition, display and processing

In view of the very small temperature and specific enthalpy changes associated with the planned tests, every effort was made to ensure the accuracy of the pressure, temperature, mass flow, speed and torque measurements and all instruments were repeatedly calibrated against instruments with recognised standards certificates.

Expander Measurements

Some indication of the expander performance is given in Figs 8 and 9.

Graphical display of results permits an output variable to be shown as a function of two input variables. This is sufficient for unambiguous presentation of single-phase

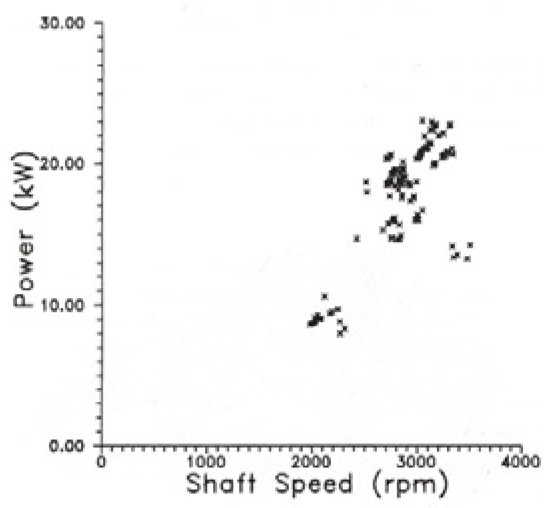

Fig. 8 Expander power output variation with rotational speed

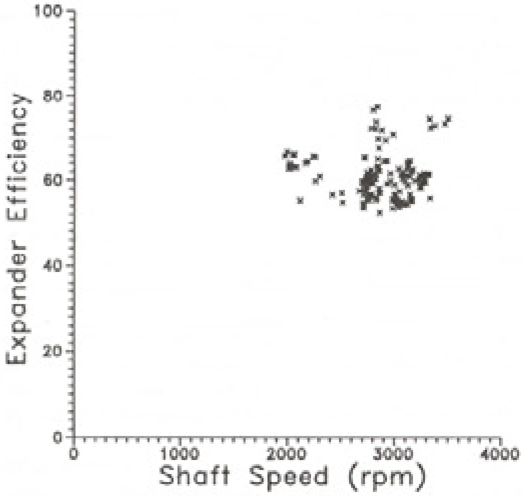

Fig. 9 Expander adiabatic efficiency variation with rotational speed

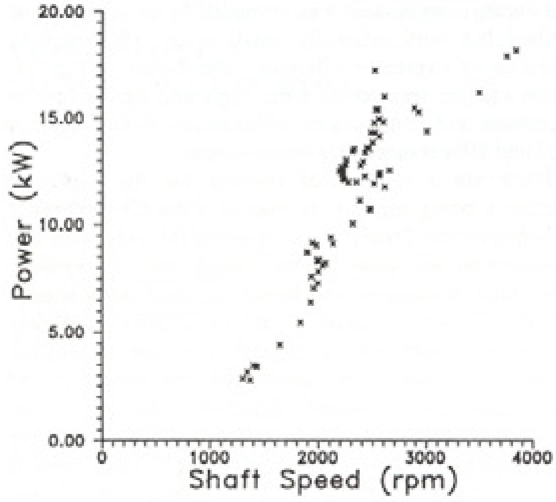

Fig.10 Expressor power output variation with rotational speed

expansion processes of fixed pressure ratio. In the case of two-phase expansion there is an additional variable of inlet fluid quality, which is not displayed. Also it was extremely difficult to control all the variables while maintaining a constant pressure ratio which did, in fact vary. These factors are the main cause of the apparent scatter of the test points.

As may be seen, a substantial number of test points were obtained at adiabatic efficiencies of over 70%. These amounted to approximately 28% of the total. To the best of the authors' knowledge these are the highest values ever achieved in any type of two-phase expander. The earlier findings of the need for a low built-in volume ratio are thus confirmed.

Expressor Measurements

Since the power produced in expansion is absorbed within the same set of rotors in an expressor, it is not possible to measure the torque and hence the power thus recovered. The main output parameter required to determine the performance is then the mass flow rate of vapour delivered by the compressor section of the expressor. This was obtained by first throttling the high pressure vapour leaving the expressor to eliminate any entrained liquid contained in it. The reduced pressure vapour was then passed through an orifice plate from which the mass flow was determined. This flow then combined with the two-phase mixture leaving the low pressure port of the expressor and returned to the condenser for recirculation after the orifice plate. The power recovery estimated from this measurement is presented in Fig. 10.

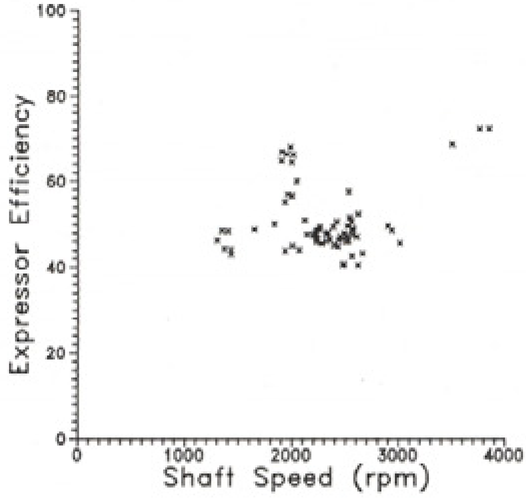

Fig.11 Expressor adiabatic efficiency variation with rotational speed

The overall efficiency of the expressor can be given as the product of its expansion and compression efficiencies. Accordingly, provided that estimates of isentropic enthalpy drop of expansion and the isentropic enthalpy rise of compression are known, the overall expressor efficiency can be determined from the mass flow measurements of the liquid input to the expressor and the vapour discharge from the high pressure port. Using R113 as the working fluid produced difficulties because it led to wet fluid discharge from the compression section and an unknown liquid carry over from the expansion process.

To overcome this uncertainty, the isentropic enthalpy rise during compression was estimated by an approximate method but with relatively small error. The resulting estimates of expressor efficiency are shown in Fig. 11. These can be seen to be quite high and correspond to expansion and compression efficiencies of the order of 70% and 80% respectively or even more.

There are a number of reasons for the expressor efficiency being superior to that of a coupled expander and compressor. Firstly, by performing the expansion and compression processes within sealed unit, no external drive shaft is required and hence the shaft seals needed for a separate expander and compressor and any mechanical losses associated with them are eliminated. Secondly, the axial forces associated with expansion and compression act in opposite directions. The unit is thus balanced and the mechanical losses associated with the thrust bearings are negligible. In addition, since there is only one set of bearings in the expressor, the losses associated with them can not be attributed to both processes. Hence, if these are allocated to the expansion process, then the vapour recompression may be assumed to take place without any effective mechanical losses or vice versa. Finally, the kinetic energy imparted to the vapour at the end of expansion is retained at the start of compression, thereby reducing the fluid dynamic losses associated with the compression process.

CONCLUSIONS

The study has demonstrated that a screw expander is a viable and stable device for use as a throttle valve replacement in large vapour compression chiller systems which can produce improvements in the Coefficient of Performance of the order of 10% at the design operating conditions. The expander can be coupled to an electrical generator or to the main compressor unit or to another compressor. Furthermore, a single pair of rotors can be used for simultaneous two-phase expansion and recompression of part of the residual vapour.

An experimental expander and later an expressor, containing only a single rotor pair, have been designed, built and tested. The results show that the concept of combining expansion and compression in one set of rotors is a valid one and that the machine designed meet the flow and pressure requirements quite closely. The machines run well and the already demonstrated ability of City "N" profile rotors to run without timing gear or oil lubrication has been upheld even when both surfaces of the rotors are used to provide a seal against internal fluid leakage. There are positive indications that the efficiency may be higher than was first thought. Since the expressor is not connected mechanically to the main compressor, an expressor may be readily incorporated into a chiller unit even as a retrofit device.

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